TANTA UNIVERSITY
FACULTY OF ENGINEERING
DEPARTMENT OF MECHANICAL POWER ENGINEERING
SECOND YEAR STUDENTS OF MECHANICAL POWER ENGINEERING

COURSE TITLE: HEAT TRANSFER (1) COURSE CODE: MEP2206

DATE:MAY, 26 SECOND TERM TOTAL ASSESSMENT MARKS: 90 TIME ALLOWED : 3 HOURS

Remarks: (answer the following questions; assume any missing data, steam and heat tables and charts are allowed)

Problem number (1) **(18 Marks)**

a) Consider two surface pressed against each other. Now the air at the interface is evacuated. Will the thermal contact resistance at the interface increase or decrease as a result? (4 Marks)

b) What is a conduction shape factor? How is it related to the thermal resistance? (4 Marks)

c) A steam pipe of inner and outer diameters 1.6 and 1.7 cm respectively is covered by with two layers of insulation. The thickness of the first layer is 3 cm and that of the second layer is 5 cm. the thermal conductivity's of the pipe and insulating layers are 58, 0.174 and 0.093 W/m.K, respectively. The temperature of the inner surface of steam pipe is 300 °C, and that of the outer surface of the insulation layer is 50 °C. Determine the heat loss per meter and the layers contact temperatures. (10 Marks)

Problem number (2) **(18 Marks)**

a) A pipe is insulated to reduce the heat loss from it. However, measurements indicate that the rate of heat loss has increased instead of decreased. Can the measurements be right? (6 Marks)

b) Steam flows through tubes whose outer diameter is 5 cm and whose walls are maintained at a temperature of 180 °C. Circular aluminum alloy fins (k = 186 W/m. °C) of outer diameter 6 cm and constant thickness 1mm are attached to the tube. The space between the fins is 3 mm, and thus there are 250 fins per meter length of the tube. Heat is transferred to the surrounding air at T_∞ = 25 °C, with a heat transfer coefficient of 40 W/m^2 . °C. determine the increase in heat transfer from the tube per meter of its length as a result of adding fins. (12 Marks)

Problem number (3) **(18 Marks)**

a) In what medium is the lumped system analysis more likely to be applicable: in water or in air? Why? (6 Marks)

b) A cubic block whose sides are 5 cm long is initially at 20 °C and are made of granite (k = 2.5 W/m. °C and α = 1.15 x 10^{-6} m^2/s) exposed to hot gases at 500 °C in a furnace on all of their surfaces with a heat transfer coefficient of 40 W/m^2. °C determine the center temperature of the cubic after 10 min. (12 Marks)

P.T.O.

TANTA UNIVERSITY
FACULTY OF ENGINEERING
DEPARTMENT OF MECHANICAL POWER ENGINEERING
SECOND YEAR STUDENTS OF MECHANICAL POWER ENGINEERING
COURSE TITLE: HEAT TRANSFER (1) COURSE CODE: MEP2206
DATE:MAY, 26 SECOND TERM TOTAL ASSESSMENT MARKS: 90 TIME ALLOWED : 3 HOURS

Remarks: (answer the following questions; assume any missing data, steam and heat tables and charts are allowed)

Problem number (1) **(18 Marks)**

a) Consider two surface pressed against each other. Now the air at the interface is evacuated. Will the thermal contact resistance at the interface increase or decrease as a result? (4 Marks)

b) What is a conduction shape factor? How is it related to the thermal resistance? (4 Marks)

(a) When two surface pressed against each other. When the air at the interface is evacuated ⇒ The thermal contact resistance at the interface will <u>increased</u> as a result. because
⇒ In the case of no fluid (an evacuated interface) eliminates conduction across the gap, thereby increasing the contact resistance
⇒ Evacuating the space between two surfaces completely eliminates heat transfer by conduction or convection but leaves the door wide open for radiation

(b) Conduction shape factor: In many instances, two- or three-dimensional conduction problems may be rapidly solved by utilizing existing solutions to the heat diffusion equation. These solutions are reported in terms of a shape factor S or a steady-state dimensionless conduction heat rate.

The shape factor is defined such that $q = S' K \Delta T_{1-2}$

and

The two-dimensional conduction resistance may expressed as

$$R_{t,cond(2D)} = \frac{1}{S' \cdot K} \quad \text{or} \quad S' = \frac{1}{K \cdot R_{th}}$$

or

In a two-dimensional system where only two temperature limits are involved, we may defined a conduction shape factor S' ⇒ $q = K.S.\,\Delta T_{overall}$

a) A steam pipe of inner and outer diameters 1.6 and 1.7 cm respectively is covered by with two layers of insulation. The thickness of the first layer is 3 cm and that of the second layer is 5 cm. the thermal conductivity's of the pipe and insulating layers are 58, 0.174 and 0.093 W/m.K, respectively. The temperature of the inner surface of steam pipe is 300 °C, and that of the outer surface of the insulation layer is 50 °C. Determine the heat loss per meter and the layers contact temperatures. (10 Marks)

ata:- $r_1 = 0.008$ m $K_1 = 58$ W/m.K

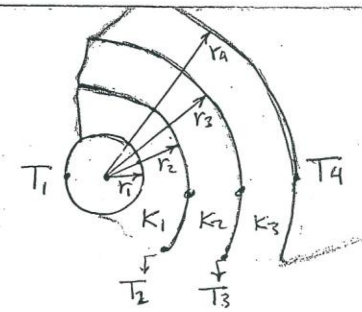
$r_2 = 0.0085$ m $K_2 = 0.174$ W/m.K

$r_3 = r_2 + 0.03 = 0.0385$ m

$r_4 = r_3 + 0.05 = 0.0885$ m

$K_3 = 0.093$ W/m.K

$T_1 = 300°C$ $T_4 = 50°C$



eq:- ① Q/ℓ ② T_2 & T_3

$$T_1 \xrightarrow{\frac{\ln r_2/r_1}{2\pi K_1 \ell}} T_2 \xrightarrow{\frac{\ln r_3/r_2}{2\pi K_2 \ell}} T_3 \xrightarrow{\frac{\ln r_4/r_3}{2\pi K_3 \ell}} T_4$$

oln

$$Q/\ell = \frac{\Delta T}{\sum R_{th}}$$

$$\Delta T = T_1 - T_4 = 250°C \quad \text{and} \quad \sum R_{th} = \frac{\ln r_2/r_1}{2\pi K_1} + \frac{\ln r_3/r_2}{2\pi K_2} + \frac{\ln r_4/r_3}{2\pi K_3}$$

$$\sum R_{th} = \frac{0.0606}{2\pi * 58} + \frac{1.5106}{2\pi * 0.174} + \frac{0.83234}{2\pi * 0.093} \simeq 2.80631$$

$$Q/\ell = \frac{\Delta T}{\sum R_{th}} = \frac{250}{2.80631} \simeq 89.085 \text{ W/m}$$

or steady state Q/ℓ is the same for (throug) all layers

$$Q/\ell = 89.085 = \frac{T_1 - T_2}{\frac{\ln r_2/r_1}{2\pi K_1}} \Rightarrow T_2 = T_1 - 89.085 * \frac{\ln r_2/r_1}{2\pi K_1}$$

$$\underline{T_2} = 300 - 89.085 * \frac{\ln(0.0085/0.008)}{2\pi * 58} = \underline{299.9852\ °C}$$

$$Q/\ell = 89.085 = \frac{T_3 - T_4}{\frac{\ln r_4/r_3}{2\pi K_3}} \Rightarrow T_3 = T_4 + 89.085 * \frac{\ln r_4/r_3}{2\pi K_3}$$

$$\underline{T_3} = 50 + 89.085 * \frac{\ln(0.0885/0.0385)}{2\pi * 0.093} = \underline{176.895\ °C}$$

Problem number (2) **(18 Marks)**

a) A pipe is insulated to reduce the heat loss from it. However, measurements indicate that the rate of heat loss has increased instead of decreased. Can the measurements be right? (6 Marks)

b) Steam flows through tubes whose outer diameter is 5 cm and whose walls are maintained at a temperature of 180 °C. Circular aluminum alloy fins (k = 186 W/m. °C) of outer diameter 6 cm and constant thickness 1mm are attached to the tube. The space between the fins is 3 mm, and thus there are 250 fins per meter length of the tube. Heat is transferred to the surrounding air at T_∞ = 25 °C, with a heat transfer coefficient of 40 W/m^2 . °C. determine the increase in heat transfer from the tube per meter of its length as a result of adding fins. (12 Marks)

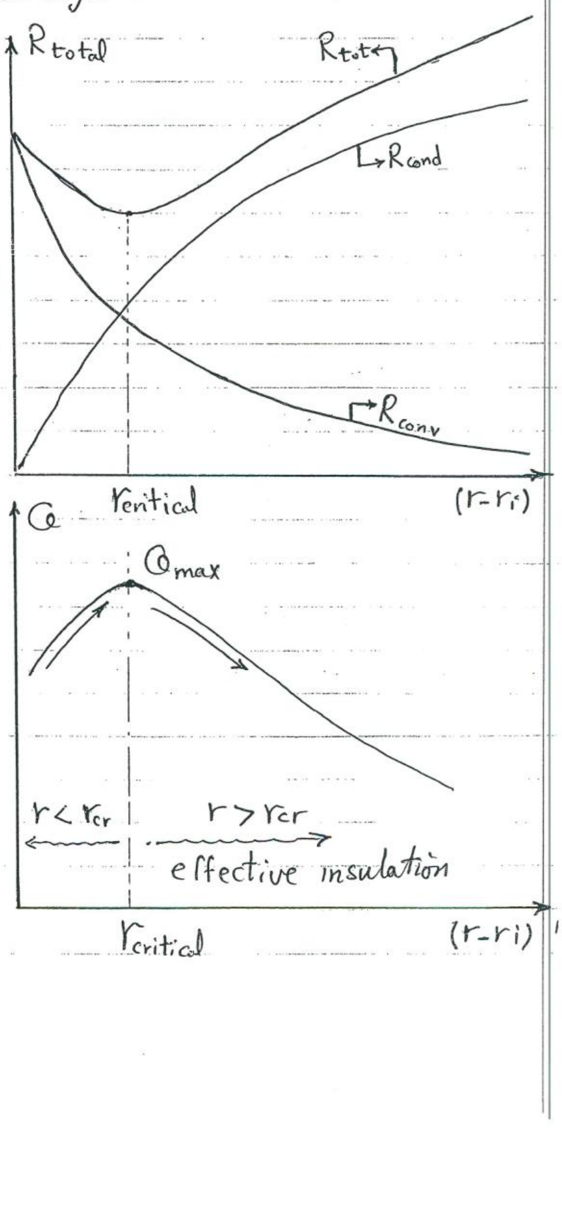
(a) yes the measurements can be right

→ In the case of $r_{cr} > r_i$ the heat transfer will increase with the addition of insulation up to a thickness of $r_{cr} - r_i$ where the total resistance decreases and the heat transfer rate therefore increase with the addition of insulation. This trend continues until the outer radius of the insulation corresponds to the critical radius.

so that

If $r < r_{cr}$ the insulation will be not effective because it causes Q increases.

but $r > r_{cr}$ the insulation will be effective because it causes Q to be decrease.



(b) $\theta_o = T_o - T_\infty = 180 - 25 = 155\,^\circ C$

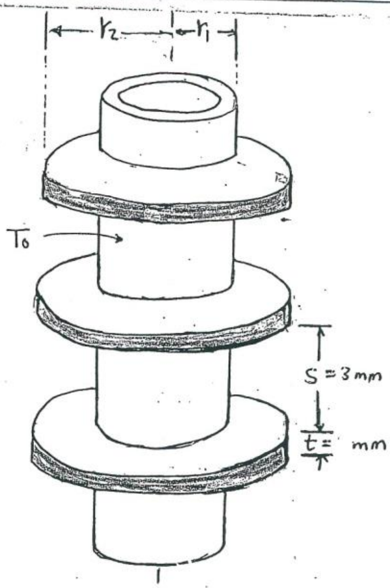
fin Thickness $(t) = 1\ mm$

Space between fins $= 3\ mm$

pipe radius $(r_1) = 2.5\ cm$

annular fin radius $(r_2) = 3\ cm$

Fin protrude $(L) = 0.5\ cm$



$\because \dot{Q}_{max} = h * \theta_o *$ Fin total area $A_{f.t}$

Where

$$A_{f.t} = 2\pi(r_2^2 - r_1^2) + 2\pi r_2 * t$$

$\therefore$

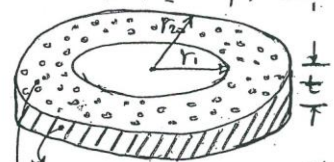
$$\dot{Q}_{max} = 40 * 155 * \left[2\pi(0.03^2 - 0.025^2) + 2\pi * 0.03 * 0.001\right]$$

$\dot{Q}_{max} = 11.8815\ W$

assume $\eta_{fin} = 0.95$ and $\eta_{fin} = \dfrac{\dot{Q}_{fin}}{\dot{Q}_{max}}$

$\dot{Q}_{fin} = 0.95 * \dot{Q}_{max} = 11.28743\ W$

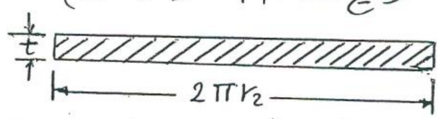
Fin total area [مساحة سطح الحلقة الزعنفة]



$= 2\pi r_2 * t \longrightarrow$ مساحة ①

$+ 2\pi(r_2^2 - r_1^2) \longrightarrow$ مساحة ②

توضيح لمساحة القشرة (مساحة ①)



$\Rightarrow$ The heat transfer from the tube surface between two adjacent fins (space between fins) $\Rightarrow \dot{Q}_{un\,fin}$

$$\dot{Q}_{un\,fin} = h * \theta_o * A_{\text{tube surface between two adjacent fins}}$$

$$= 40 * 155 * (2\pi r_1 * S)$$

$$= 40 * 155 * (2\pi * 0.025 * 0.003)$$

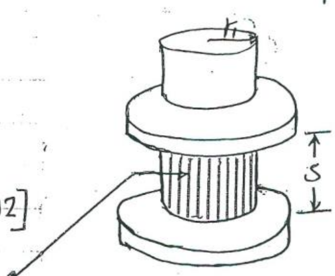
$\dot{Q}_{un\,fin} = 2.9217\ W$

Where the number of fins = 250 fins/ 1m

So that the spacing between fins also = 250

$\Rightarrow \dot{Q}_{Total\,(fins + spacing)} = 250\,[11.287 + 2.92]$

$\dot{Q}_{Total} = 3552.278\ W$



Area of tube surface between two adjacent fins

Where the required is the increase in heat transfer from the tube per meter of its length as a result of adding fins.
we firstly calculate $\dot{Q}_{no,fin}$ where

$$\dot{Q}_{no,fin} = h * \theta_o * (2\pi r_1 L) = 40 * 155 * (2\pi * 0.025 * 1)$$

$$\dot{Q}_{no,fin} = 973.894 \text{ W}$$

So that

$$\dot{Q}_{increased} = \dot{Q}_{total(fins+spacing)} - \dot{Q}_{no,fin}$$
$$= 3552.278 - 973.894$$
$$\dot{Q}_{increased} = 2578.384 \text{ W}$$

$$\epsilon_{overall} = \frac{\dot{Q}_{Total}}{\dot{Q}_{no,fin}} = \frac{3552.278}{973.894} = 3.6475$$

Problem number (3) **(18 Marks)**

a) In what medium is the lumped system analysis more likely to be applicable: in water or in air? Why? (6 Marks)

b) A cubic block whose sides are 5 cm long is initially at 20 °C and are made of granite (k = 2.5 W/m. °C and α = 1.15 x 10^{-6} m²/s) exposed to hot gases at 500 °C in a furnace on all of their surfaces with a heat transfer coefficient of 40 W/m². °C determine the center temperature of the cubic after 10 min. (12 Marks)

(a) Where the basic condition for the lumped system analysis to be applicable is $Bi = \frac{h * L_c}{k} \leq 0.1$

and this condition ($Bi \leq 0.1$) is more likely to be applicable when "h" is small which achieved in air

Where $h_{air} < h_{water}$

so that " The lumped system analysis is more likely to be applicable in air "

(b) Data: A cubic block $L = 5$ cm $T_i = 20°C$
$K = 2.5$ W/m.°C and $\alpha = 1.15 * 10^{-6}$ m²/s , $T_\infty = 500°C$
$h = 40$ W/m².°C , τ (time) = 10 min req:- $T(0,0,0,\tau) = ?°C$

The cubic block can physically be formed by the intersection of three infinite plane walls of Thickness $2L = 5\,cm$. $\Rightarrow L = 2.5\,cm$

After 10 minutes:

$$Bi = \frac{h*L}{K} = \frac{40 * 0.025}{2.5} = 0.4$$

$$Fo = \frac{\alpha \cdot \tau}{L^2} = \frac{1.15*10^{-6} * 10*60}{(0.025)^2} = 1.104$$

Where $Bi = 0.4 > 0.1$ and $Fo > 0.2$

so that We solve the unsteady system by using heisler chart

From chart No.1 for plane wall where $\frac{1}{Bi} = 2.5$, $Fo = 1.104$

$$\frac{\theta_o}{\theta_i} = \frac{T_o - T_\infty}{T_i - T_\infty} \simeq 0.75 = \theta(0,\tau)_{wall}$$

from the product solution for the cubic block as 3D

$$\theta(0,0,0,\tau) = \theta(x,\tau) \cdot \theta(y,\tau) \cdot \theta(z,\tau)$$
$$= \left[\theta(0,\tau)_{wall}\right]^3 = (0.75)^3 = 0.421875$$

$$\theta(0,0,0,\tau) = \frac{T(0,0,0,\tau) - T_\infty}{T_i - T_\infty} = 0.421875$$

$$T(0,0,0,\tau) = 0.421875(20-500) + 500$$

$$T(0,0,0,\tau) = 297.5\,^\circ C$$

$\Rightarrow$ The center temperature of the cubic after 10 min from the start of heating $\simeq 300\,^\circ C$

Problem number (4) **(18 Marks)**

a) What is meant by a time constant ? (4 Marks)

b) Define the reradiating surface, radiation shields, and irradiation. (4 Marks)

problem number (4) (c) ⑦

EXAMPLE 13.5

A cryogenic fluid flows through a long tube of 20-mm diameter, the outer surface of which is diffuse and gray with $\varepsilon_1 = 0.02$ and $T_1 = 77\,\text{K}$. This tube is concentric with a larger tube of 50-mm diameter, the inner surface of which is diffuse and gray with $\varepsilon_2 = 0.05$ and $T_2 = 300\,\text{K}$. The space between the surfaces is evacuated. Calculate the heat gain by the cryogenic fluid per unit length of tubes. If a thin radiation shield of 35-mm diameter and $\varepsilon_3 = 0.02$ (both sides) is inserted midway between the inner and outer surfaces, calculate the change (percentage) in heat gain per unit length of the tubes.

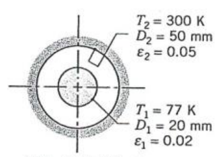
SOLUTION

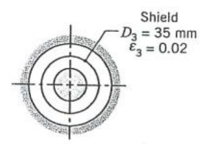
Known: Concentric tube arrangement with diffuse, gray surfaces of different emissivities and temperatures.

Find:

1. Heat gain by the cryogenic fluid passing through the inner tube.
2. Percentage change in heat gain with radiation shield inserted midway between inner and outer tubes.

Schematic:





Assumptions:

1. Surfaces are diffuse and gray and characterized by uniform irradiation and radiosity.
2. Space between tubes is evacuated.
3. Conduction resistance for radiation shield is negligible.
4. Concentric tubes form a two-surface enclosure (end effects are negligible).

Analysis:

1. The network representation of the system without the shield is shown in Figure 13.11, and the heat rate may be obtained from Equation 13.25, where

$$q = \frac{\sigma(\pi D_1 L)(T_1^4 - T_2^4)}{\dfrac{1}{\varepsilon_1} + \dfrac{1-\varepsilon_2}{\varepsilon_2}\left(\dfrac{D_1}{D_2}\right)}$$

Hence

$$q' = \frac{q}{L} = \frac{5.67 \times 10^{-8}\,\text{W/m}^2\cdot\text{K}^4\,(\pi \times 0.02\,\text{m})[(77\,\text{K})^4 - (300\,\text{K})^4]}{\dfrac{1}{0.02} + \dfrac{1-0.05}{0.05}\left(\dfrac{0.02\,\text{m}}{0.05\,\text{m}}\right)}$$

$$q' = -0.50\,\text{W/m}$$ ◁

8

2. The network representation of the system with the shield is shown in Figure 13.12, and the heat rate is now

$$q = \frac{E_{b1} - E_{b2}}{R_{\text{tot}}} = \frac{\sigma(T_1^4 - T_2^4)}{R_{\text{tot}}}$$

where

$$R_{\text{tot}} = \frac{1-\varepsilon_1}{\varepsilon_1(\pi D_1 L)} + \frac{1}{(\pi D_1 L)F_{13}} + 2\left[\frac{1-\varepsilon_3}{\varepsilon_3(\pi D_3 L)}\right] + \frac{1}{(\pi D_3 L)F_{32}} + \frac{1-\varepsilon_2}{\varepsilon_2(\pi D_2 L)}$$

or

$$R_{\text{tot}} = \frac{1}{L}\left\{\frac{1-0.02}{0.02(\pi \times 0.02\text{ m})} + \frac{1}{(\pi \times 0.02\text{ m})1} + 2\left[\frac{1-0.02}{0.02(\pi \times 0.035\text{ m})}\right] + \frac{1}{(\pi \times 0.035\text{ m})1} + \frac{1-0.05}{0.05(\pi \times 0.05\text{ m})}\right\}$$

$$R_{\text{tot}} = \frac{1}{L}(779.9 + 15.9 + 891.3 + 9.1 + 121.0) = \frac{1817}{L}\left(\frac{1}{\text{m}^2}\right)$$

Hence

$$q' = \frac{q}{L} = \frac{5.67 \times 10^{-8}\text{ W/m}^2\cdot\text{K}^4\,[(77\text{ K})^4 - (300\text{ K})^4]}{1817\text{ (1/m)}} = -0.25\text{ W/m} \quad \triangleleft$$

The percentage change in the heat gain is then

$$\frac{q'_w - q'_{wo}}{q'_{wo}} \times 100 = \frac{(-0.25\text{ W/m}) - (-0.50\text{ W/m})}{-0.50\text{ W/m}} \times 100 = -50\% \quad \triangleleft$$

Comment: Because the geometries are concentric and the specified emissivities and prescribed surface temperatures are spatially uniform, each surface is characterized by uniform irradiation and radiosity distributions. Hence the calculated heat transfer rates would not change if the cylindrical surfaces were to be subdivided into smaller radiative surfaces.

13.3.5 The Reradiating Surface

The assumption of a *reradiating surface* is common to many industrial applications. This idealized surface is characterized by *zero* net radiation transfer ($q_i = 0$). It is closely approached by real surfaces that are well insulated on one side *and* for which convection effects may be neglected on the opposite (radiating) side. With $q_i = 0$, it follows from Equations 13.15 and 13.19 that $G_i = J_i = E_{bi}$. Hence, if the radiosity of a reradiating surface is known, its temperature is readily determined. In an enclosure, the equilibrium temperature of a reradiating surface is determined by its interaction with the other surfaces, and it is *independent of the emissivity of the reradiating surface.*

A three-surface enclosure, for which the third surface, surface R, is reradiating, is shown in Figure 13.13a, and the corresponding network is shown in Figure 13.13b. Surface R is presumed to be well insulated, and convection effects are assumed to be negligible. Hence, with $q_R = 0$, the net radiation *transfer* from surface 1 must equal the net radiation

(a) Time constant $\tau^* \equiv \tau_t = \left(\frac{1}{h A_s}\right) \cdot (\rho V C) = R_t \cdot C_t$

Where R_t is the resistance to convection heat transfer and C_t is the lumped thermal capacitance of the solid. Any increase in R_t or C_t will cause a solid to respond more slowly to changes in its thermal environment. This behavior is analogous to the voltage decay that occurs when a capacitor is discharged through a resistor in an electrical RC circuit.

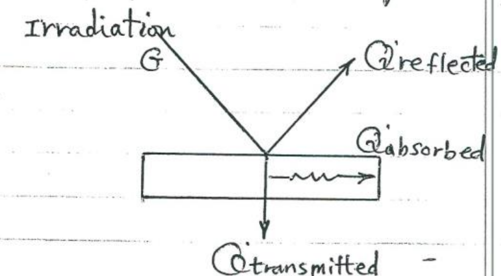
(b) * Re-radiating surface: is an idealized surface characterized by zero net radiation transfer ($q_i = 0$). It is closely approached by real surfaces that are well insulated on one side and for which convection effect may be neglected on the opposite (radiating) side. and for re-radiating surface $J_i = \sigma T_i^4$

* Radiation shield: Radiation heat transfer between two surfaces can be reduced greatly by inserting a thin, high-reflectivity (low-emissivity) sheet of material between two surfaces. such high reflectivity thin plates or shells are called radiation shields. $\dot{q}_{rad} \rightarrow$ decreased $\Leftrightarrow \Sigma R_{th} \rightarrow$ increased

* Irradiation: The radiation flux incident on a surface from all directions (G) or Irradiation represents the rate at which radiation energy is incident on a surface per unit area of the surface where

$G = \dot{Q}_{ref} + \dot{Q}_{abso} + \dot{Q}_{trans}$

↳ The total incident radiation



Problem number (5) **(18 Marks)**

a) Explain the Planck distribution for the black body? (6 Marks)

b) A large uranium plate of thickness L = 6 cm and thermal conductivity k = 28 W/m. °C in which heat is generated uniformly at a constant rate of g = 5 x 10^6 W/m^3. one side of the plate is maintained at 0 °C by iced water while the other side is subjected to convection to an environment at T_∞ = 30 °C with a heat transfer coefficient of 45 W/m^2. °C, considering a total of four equally spaced nodes in the medium, two at the middle and two at the boundaries, estimate the exposed surface temperature of the plate under steady conditions using the finite difference approach. (12 Marks)

(a) The plank distribution:

The black-body spectral intensity is well known, having first been determined by blank.

$$I_{\lambda,b}(\lambda,T) = \frac{2hC_o^2}{\lambda^5[\exp(hC_o/\lambda K_B T)-1]}$$

Where $h = 6.626 \times 10^{-34}$ J.s and $K_B = 1.381 \times 10^{-23}$ J/K are the universal Plank and Boltzman constant, respectively, $C_o = 2.998 \times 10^8$ m/s is the speed of light in Vaccum, and T is the absolute temperature of the blackbody (K).

Since the blackbody is a diffuse emitter, it follows from equation ② that its spectral emissive power is

$$E_{\lambda,b}(\lambda,T) = \pi I_{\lambda,b}(\lambda,T) = \frac{C_1}{\lambda^5[\exp(C_2/\lambda T)-1]}$$

Equ ②

The plank's law or distribution

Where the first and second radiation constant are

$C_1 = 2\pi h C_o^2 = 3.742 * 10^8$ W.μm^4/m^2 and

$C_2 = (hC_o/K_B) = 1.439 \times 10^4$ μm.K.

⇒ from the plank's distribution we note that:

1- The emitted radiation varies continuously with wavelength.

2- At any wavelength the magnitude of the emitted radiation radiation increases with increasing temperature.

3- The spectral region in which the radiation is concentrated depends on temperature, with comparatively more radiation appearing at shorter wave-lengths as the temperature increases.

4- A significant fraction of the radiation emitted by the sun which may be approximated as a blackbody at 5800 K is in the visible region of the spectrum.